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SIMULATION OF A NATURAL CONVECTION HEAT EXCHANGER SOLAR DOMESTIC HOT WATER SYSTEM

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ABSTRACT

A model of a natural convection heat exchanger (NCHE) in a solar domestic hot water (SDHW) system is presented for use with the TRNSYS program. The model, based upon crossflow heat exchanger correlations, requires only geometric specifications of the NCHE. By varying heat exchanger geometric parameters (such as the number of helices, diameters of helices, diameter and length of the heat exchanger shell) the model can be used to design an optimum NCHE. Comparisons of the model to correlations of experimental data reported by Fraser (1) show reasonable agreement.

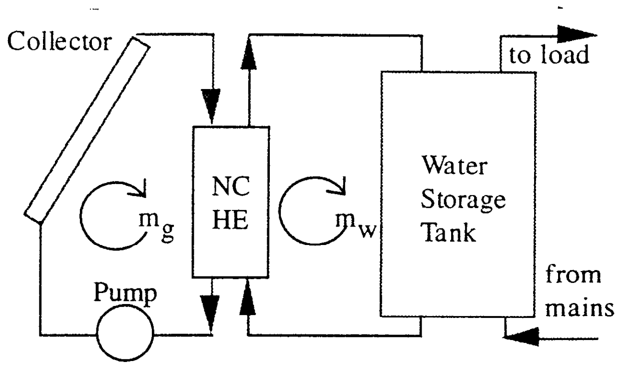

Fig. 1 Schematic of solar domestic hot water system with NCHE.

1. INTRODUCTION

Natural convection heat exchangers (NCHEs) can simplify SDHW systems in which an antifreeze loop is needed for freeze protection. As is shown in Figure 1, a NCHE is a sidearm heat exchanger in which the water is circulated by natural convection. A properly-designed SDHW system employing a NCHE offers the possibilities of good tank stratification, ease in retrofitting, and eliminates the need for a pump.

NCHE systems are difficult to design. To promote water flow, large diameter pipes and few fittings should be used to minimize pressure losses. However, no other guidelines exist for optimizing the design of an efficient NCHE system. In order to assist in designing an optimum NCHE system, simulation models have been written. Fraser (1) developed a model for a shell and coil NCHE for use in the WATSUN simulation program (2). The model is based on correlations of experimental pressure loss and heat exchanger effectiveness as a function of flowrate for a NCHE manufactured by Thermo Dynamics Ltd. of Canada. The model is presented in Fraser (1) and Bergelt (3).

2. A FIRST PRINCIPLES MODEL

A model was written to simulate NCHE performance without reliance on experimental data. This model uses correlations found in the literature and it should be applicable to any shell and coil geometric configuration. This model, designed for use with TRNSYS (4), is useful for investigating the effect of heat exchanger geometric parameters on system performance. By varying the geometry of the heat exchanger in the model, the model can be used as a tool for optimization of the design of a shell and coil NCHE.

2.1 Geometry of a Shell and Coil NCHE

Pressure drop and heat transfer correlations for forced flow over helical coils do not exist although a very recent study (5) has contributed a natural convection correlation for a NCHE. As an approximation, correlations for bundles of tubes in crossflow are used. The first step therefore, is the modification of the geometry of the coils such that they can be represented as bundles of tubes.

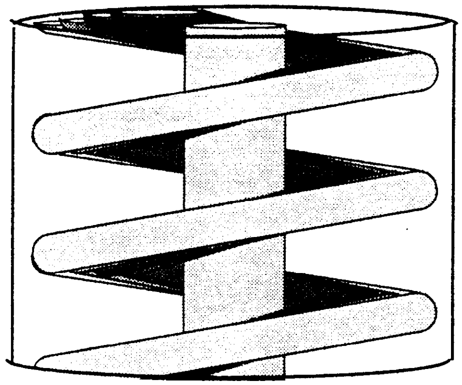

Fig. 2 Inside side-view of shell and coil heat exchanger. Glycol flows within the coils, water flows over the coils.

Figure 2 presents a cutaway of a shell and coil NCHE. For every pitch length, one complete revolution of each coil is exposed to crossflow. Hence, if the slope of the tubes are neglected, the coils have nearly the same pressure drop as a series of toroids of the same diameter. These toroids are then cut and straightened, so as to resemble straight tubes in crossflow. If there are four helical coils, there now will be four columns of straight tubes, each of different length. An average of these lengths is taken so that the bundle of tubes now can be represented as a bundle of tubes in crossflow for which heat transfer and pressure drop correlations are available.

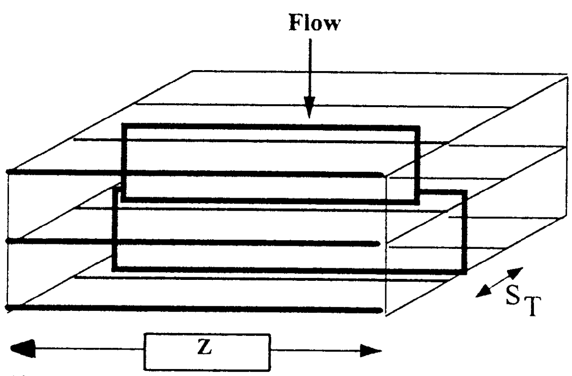

Fig. 3 Helical coils represented as tubes in crossflow

Figure 3 depicts the transformed coils as a bundle of tubes in crossflow. The coil pitch is S_L, and the average distance between the coils in the transverse direction is S_T. The number of tube rows in the longitudinal direction is found using:

$$N_L = \frac{H}{S_L} \tag{1}$$

where H represents the height of the heat exchanger. The number of tube columns equals the number of coils in the original configuration, hence:

$$N_T = N_{coils} \tag{2}$$

The depth, Z, is found by averaging the lengths of the tubes:

$$Z = \frac{\sum_{i}^{N_{coils}} \pi D_c}{N_T} \tag{3}$$

where D_C is the coil diameter. The tube outer surface area is found using:

$$A_{s,o} = \pi D_{t,o} Z N \tag{4}$$

where $D_{t,o}$ is the tube outer diameter and N is the total number of tubes.

2.2 Pressure Drop Model

In a natural convection system, pressure differences are what determine the water flow rate. Minor losses result from 45° and 90° ells and pipe entrance and exit conditions. The minor losses are accounted for in the model with accepted K-factor values for the entrance and exit conditions and with Hooper's correlation (8) for the 45° and 90° ells. The shear pressure drop is found with a correlation presented by Jakob (6):

$$\Delta P_{sh} = - f_j N \left(\frac{1}{2} \rho_{w,i} U_{max}^2\right) \cdot \left(\frac{\mu_s}{\mu_{w,i}}\right)^{0.14} \tag{5}$$

where Jakob's friction factor is found using:

$$f_j = \left[0.176 + \frac{0.34\, S_L / D}{\left(S_T / D - 1\right)^{0.43 + D / S_T}}\right] \cdot \mathrm{Re}_{D,\max}^{-0.15} \tag{6}$$

and where: $\rho_{w,i}$ = water density at the inlet, U_{max} = maximum water velocity, μ_i= water viscosity at average tube surface temperature and $\mu_{w,i}$ = water viscosity the inlet. Properties are evaluated at the average water temperature, except for $\mu_{w,i}$ and $\rho_{w,i}$, which are evaluated at the water inlet temperature, and μ_s, which is evaluated at the average tube surface temperature. Jakob's pressure drop correlation requires a Reynolds number, $Re_{D,max}$, which is based upon the maximum fluid velocity, μ_{max} (which occurs in the transverse flow area), and the diameter of the tubes in crossflow, $D_{t,o}$.

$$Re_{D,\max} = \frac{U_{\max} D_{t,o}}{\upsilon} \tag{7}$$

The kinematic viscosity, υ, is evaluated at the average heat exchanger water temperature.

Figure 4 presents a comparison of the pressure drop calculated with the model to correlations of experimental shear pressure drop presented by Fraser (1). The Jakob analysis resulted in errors averaging 3.3% over the range shown.

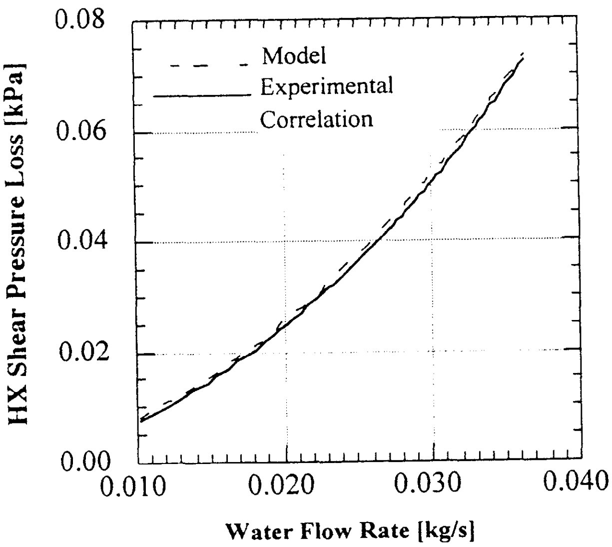

Fig. 4 Comparison of simulated NCHE shear pressure loss to Fraser's correlations (1).

$$C^* = \frac{(\dot{m}\, C_p)_g}{(\dot{m}\, C_p)_w}$$

2.3 Heat Transfer Model

An effectiveness-NTU approach is used to simulate the heat transfer in the NCHE. The UA_S value of the heat exchanger is found using:

$$U\, A_S = \frac{1}{\frac{1}{h_i A_{S,i}} + \frac{1}{h_o A_{S,o}}} \tag{8}$$

where $A_{s,o}$, $A_{s,i}$, h_o, and h_i, are the outer and inner tube surface areas, and the outer and inner heat transfer coefficients respectively. The resistance of the copper wall is assumed to be negligible. The number of transfer units is defined as:

$$NTU = \frac{UA_s}{(\dot{m}\, C_p)_g} \tag{9}$$

The effectiveness can then be found using:

$$\varepsilon' = \frac{1-\exp\left[-NTU\,(1-C^*)\right]}{1-C^*\exp\left[-NTU\,(1-C^*)\right]} \tag{10}$$

where:

$$C^* = \frac{(\dot{m}\, C_p)_g}{(\dot{m}\, C_p)_w} \tag{11}$$

The definition of the capacitance rate ratio in Eqn. 11 remains unchanged regardless of which stream has the minimum capacitance rate. For this reason, ε' in Eqn. 10 is called a modified effectiveness.

No heat transfer correlations could be found for the heat transfer associated with *forced* flow over helical tubes. Consequently, Zukauskas' (7) correlation for flow over tube bundles was used to find an average heat transfer coefficient.

$$\overline{Nu}_o = C\, Re^m_{D,\max}\, Pr^n \left(\frac{Pr}{Pr_s}\right)^{1/4} \tag{12}$$

where Pr was evaluated at the average heat exchanger water temperature, Pr_s, at the average tube surface temperature and where the constants C, m and n, varied depending upon the Reynolds number as is shown in Table 1.

TABLE 1. ZUKAUSKAS' PARAMETERS

$Re_{D,max}$	C	m	n
0 - 100	0.9	0.4	0.36
100 - 1000	0.52	0.5	0.36

Reynolds numbers range in this analysis from 0 to 150. As calculations made using both sets of parameters in Table 1 gave nearly the same results, either set of parameters would suffice. The coefficients applying to the lower range of Reynolds numbers were chosen.

Zukauskas' correlation has a 15% uncertainty. More inaccuracy may result from this particular analysis, as flow over the helical coils may produce a swirling motion which may enhance heat transfer.

Heat transfer is greater in helical tubes than in straight tubes due to the secondary flows that are established in the tubes. Flow tends to be more irregular, with higher fluid velocities at the outer tube wall and lower fluid velocities at the inner wall. The higher fluid velocities considerably decrease thermal resistance in the fluid, thereby yielding greater heat transfer coefficients in helical tubes. Manlapanz and Churchill's correlation (8) for flow in helical tubes was used to find the Nusselt number for the glycol flow in an average coil, in which constant heat flux was assumed.

$$Nu_i = \left[\left(4.364 + \frac{4.636}{x_1}\right)^3 + 1.816\left(\frac{De}{x_2}\right)^{3/2}\right]^{1/3} \quad (13)$$

where:

$$x_1 = \left(1 + \frac{1352}{De^2 \Pr}\right)^2 \quad \text{and} \quad x_2 = 1 + \frac{1.15}{\Pr}$$

The Dean number, a non-dimensionalized parameter which accounts for secondary flow in the helix, is found using:

$$De = \mathrm{Re}\sqrt{\frac{a}{R}} \quad (14)$$

where a represents the tube radius, and R represents the radius of curvature of the helix.

For glycol inlet temperatures below 80°C, the detailed model predicts a heat transfer rate that is up to 15% greater than that from by correlations of experimental data provided by Fraser (1).

3. COMPARISON OF MODELS

One year simulations for Madison were performed using the model described in this paper and a model similar to that of Fraser developed for TRNSYS (4). The annual solar fractions were 51% for the Fraser model and 52% for the model described in this paper. By using Zukauskas' crossflow correlation, the enhanced heat transfer due to the swirling motion of water over helical coils was neglected. This conservative assumption should lead to an slight underprediction of the heat transfer in the heat exchanger but this was not observed.

4. NCHE OPTIMIZATION

The parameters which affect the performance of an NCHE system can be grouped into heat exchanger and system parameters. The heat exchanger parameters are: 1) number of coils; 2) heat exchanger shell length; 3) tube diameter; 4) tube spacing; and 5) heat exchanger shell diameter. The system parameters are: 1) location (weather data), 2) glycol flow rate, 3) collector array size; and 4) the water draw.

TRNSYS simulations were performed on various NCHE geometries while keeping system parameters constant. Each simulation provides the fraction of the load supplied by solar energy which was used in conjunction with an economic analysis to determine the best heat exchanger design. A range of heat exchanger geometries was then subjected to variations in system parameters in order to determine the effect of each of these parameters upon the life-cycle savings. The base-case conditions and parameter settings were:

Weather Data:
TRNSYS weather generator: Madison, WI

Collector Specifications (SRCC (9) 1993):
$F_R U_{L1}$ = 4.02 W/m^2-°C (2nd order fit)
$F_R U_{L2}$ = 0.0072 W/m^2-°C^2
$F_R \tau\alpha$ = 0.634
b_o=0.448 b_1=-0.234
Test flow rate: 11.86 kg/hr-m^2
Collector area: 4.5 m^2

Tank: Tank Volume: 454 L
5 node stratified tank
Daily DHW Draw: 260 L
Demand Profile Fig. 5

Piping: Fig. 6 and Table 2.

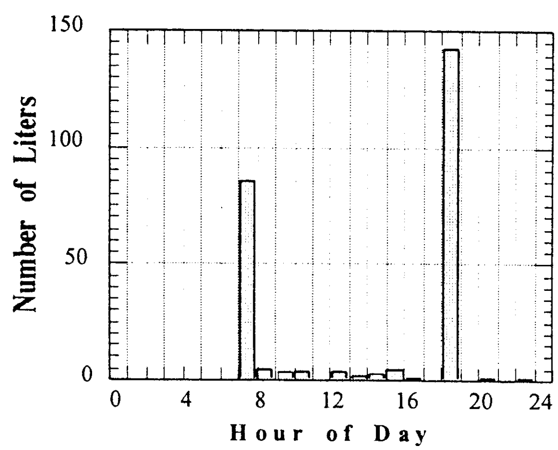

Fig. 5 Average daily hot water draw profile WATSIM (10).

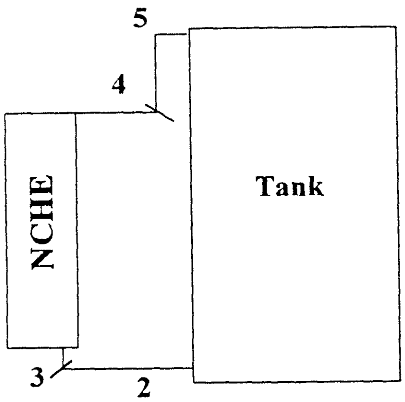

Fig. 6 Diagram of pipe locations.

In designing an optimum shell and coil heat exchanger, economic considerations need to be considered. The potential increased cost of purchasing a larger or more intricate heat exchanger must be balanced with the additional yearly savings possible from an improved design. Frequently domestic hot water systems can be improved to promote system efficiency, but at considerable cost in hardware, that such improvements are not economically justified. The economic analysis used in weighing the heat exchanger cost against the projected yearly fuel savings employs the life cycle savings (LCS) method in conjunction with the P_1, P_2 method described by Duffie and Beckman (11).

TABLE 2. PIPING PARAMETERS

Pipe Location	Length [m]	Vertical Rise [m]	Diameter [m]	K
2	0.5	0	0.01905	1.5
3	0.5	0.0635	0.01905	1.5
4	0.5	0	0.01905	1.5
5	0.5	0.965	0.01905	1.5

The economic optimization assumes a 10 year economic analysis. Maintenance and parasitic costs are considered negligible, there are no assumed property taxes on the equipment, and the resale value after 10 years is zero. The down payment is 1/6 of the equipment cost, while the rest is paid in a 5 year mortgage with a 9.5% interest rate. The inflation rate of fuel is assumed to be 6.5%. The discount rate is taken as 10.5%. The owner's effective tax bracket is assumed to be 0.42. Using these parameters, P_1 is found to be 7.709 and P_2 is 0.894.

The objective of the optimization is to increase the life-cycle savings of the a SDHW system employing the an NCHE similar to the Thermo Dynamics shell and coil heat exchanger. The heat exchanger cost was then estimated for different lengths and number of coils using the following assumptions:

1) Every heat exchanger will have a fixed cost associated with it for headers, inlet and outlet piping, and heat exchanger shell. This cost, C_{fixed}, is assumed to be \$100.00.

2) The labor involved to bend one coil of 0.4064 m would cost \$10.00. The labor cost can therefore be represented by:

$$C_{labor} = \frac{\$10.00}{coil} N_{coils} \frac{L_{HX}}{0.4064\ m} \tag{15}$$

where L_{HX} is the heat exchanger shell length.

3) The tubing cost per heat exchanger is:

$$C_{tubing} = \frac{\$1.97}{m} L_{tubing} \tag{16}$$

where L_{tubing} is the total tube length.

4) The manufacturer charges twice the manufacturing cost for the heat exchanger.

The cost of the 0.4064 m, 4 coil, shell and coil NCHE is approximately \$400.00 based on these assumptions. Additional details are provided by Avina (12).

Preliminary simulations showed that tube diameter, and tube spacing had little effect upon NCHE performance. The geometric parameters that had the greatest effect upon system performance were those relating to heat transfer area: the length of the HX shell (and consequently the total tube length), and the number of coils.

Using an S_T of 9.5 mm and 6.35 mm diameter tubes, the number of coils and the heat exchanger length were varied in order to generate a solar fraction and an estimated heat exchanger cost. From Figure 7 the optimal NCHE configuration is one with two coils each 0.45 m in length.

It was found that all of the system parameters tested had an effect upon the optimal heat exchanger design. However, in all simulation results, it was found that a 2 coil heat exchanger was economically the most beneficial. Fig 8 shows the effect of collector area and heat exchanger length for a 2-coil exchanger in Madison. The optimum heat exchanger length increases with increased collector area.

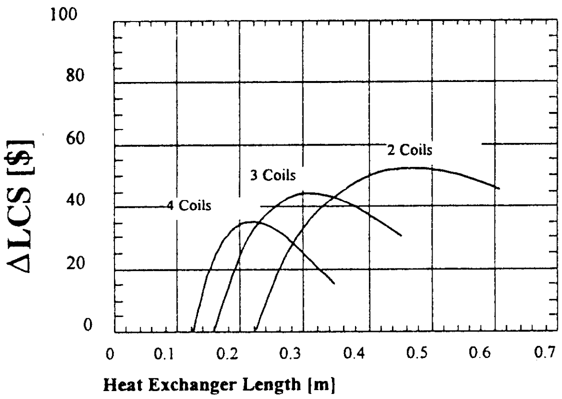

Fig. 7: ΔLCS as a function of heat exchanger length and number of coils.

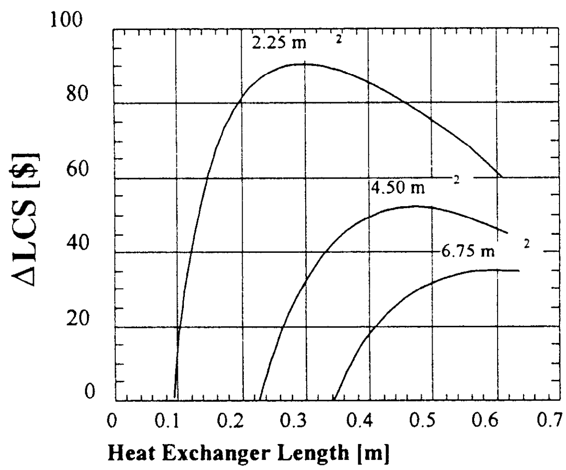

Fig. 8 The effect of collector area and heat exchanger length for a 2-coil heat exchanger design

5. CONCLUSIONS

A model for a natural convection heat exchanger in a solar domestic water heating system has been developed for TRNSYS. The model is based upon crossflow correlations found in the literature. Comparisons of the model with correlations of experimental data provided by Fraser (1) show adequate agreement. The model was used to investigate the effects of heat exchanger parameters on SDHW performance and economics. It was found that the heat exchanger size was a major parameter and that areas smaller than that provided by Thermo Dynamics Ltd. led to greater system performance. Coil spacing and tube diameter had a lesser impact upon system performance. The optimal heat exchanger design for the base case contained 2 helices and was 0.45 m long. For a given set of system parameters, a SDHW system containing the optimally designed heat exchanger would save the consumer an extra $110 in initial equipment cost, and $52 over a 10 year period. Heat exchanger designs were subject to variations in system parameters, such as collector area, hot water draw, location and glycol flow rate. Although each set of system parameters suggested a different optimal design, overall, the optimal design found for the initial set of system parameters remained adequate. As different economic assumptions will lead to differing optimal heat exchanger lengths, this paper can serve as a guide for those who desire to optimize a shell and coil NCHE based upon a prevailing set of economic assumptions.

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